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ORNL
CENTRAL FILES NUMBER

CF 58-7-35

UCC

POST OFFICE BOX X
OAK RIDGE, TENNESSEE

DATE: July 9, 1958, 1958
SUBJECT: HRT Pressure Vessel Temperatures During Operation -
An Evaluation Using Data from Run 16
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COPY NO. 99

SUMMARY

A study has been made of the data taken during HRT Run 16, concerning the temperatures and thermal stresses in the pressure vessel. By making heat balances across the blast shield cooling coils, the pressure vessel cooling system was shown to be performing satisfactorily. Estimates of the thermal stress in the pressure vessel wall show values which range from 1800 psi at low power to 6800 psi at 5 Mw. The thermal stress does not appear to be excessive in this power range. The overall heat generation in the pressure vessel was estimated to be 108,000 Btu/hr.Mw. By comparing the heat flux at the inside surface of the pressure vessel with the temperature drop across the blanket solution film, which would be required to force compliance with the theoretical equations, a heat transfer film coefficient at this surface may be estimated. A rough correlation between this film coefficient and the temperature drop across the film indicates that heat is being transmitted principally by natural convection.

It is recommended that a complete temperature profile be attempted on the replacement reactor pressure vessel. If this cannot be done, then provision for the measurement of the inside surface temperature should be made.

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Introduction

Run 16 of the HRT started on May 29, 1958, and ended with a scheduled shutdown on July 3, 1958. During this time the reactor operated successfully at various power levels starting at zero power on June 4, and ending at slightly above five megawatts on July 3. As a result of a leak between the fuel and blanket systems, the machine was operated with uranium in both regions. The uranium concentration in the core was 5.4 g U-235/kg D₂O and in the blanket was 1.88 g U-235/kg D₂O.

Run 16 represents the first extended power run of the HRT. During this run a mass of data was collected which included, along with power level:

- 1) blanket solution inlet and outlet temperatures,
- 2) outside surface temperatures of the pressure vessel, and
- 3) the blast shield cooling water's flow rate, inlet and outlet temperatures.

These data have been analyzed for the following purposes:

1. To estimate the overall heat generation rate in the pressure vessel wall.
2. To determine the effectiveness of the pressure vessel cooling system in removing a part of the heat generated in the pressure vessel wall.
3. To estimate the thermal stresses produced in the pressure vessel wall.
4. To estimate the actual heat transfer film coefficient between the inside surface of the pressure vessel and the blanket solution.

The specific purpose of the cooling system is to maintain a low temperature differential across the pressure vessel wall at the design power level. This is done to minimize the thermal stress in the pressure vessel which is related directly to the temperature differential across the wall. The design of the cooling system is given in CF 55-3-119.⁽¹⁾

Since no provision was made to measure the inside surface temperature of the pressure vessel directly, this temperature was estimated using an equation derived in CF 55-7-64.⁽²⁾ The difference between this temperature and the outside surface temperature, along with the reactor power, gives a value from which the thermal stress can be calculated.

A final purpose of this study is to recommend what further provisions are needed for the taking of data on the replacement reactor vessel.

Pressure Vessel Cooling System

Cooling of the pressure vessel during operation is accomplished by radiating heat from the outside surface across a 1-1/2" to 2" air gap to the blast shield, a concentric, spherical stainless steel shell placed around the pressure vessel. The blast shield is maintained at a temperature of about 100°F by cooling water flowing through coils welded directly on the outside surface. Estimates of the heat removed from the outside surface of the pressure vessel can be made using the equations⁽¹⁾

$$\left(\frac{q}{A}\right)_{\text{rad}} = \frac{0.173}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1} \left[\left(\frac{T_b + 460}{100} \right)^4 - \left(\frac{T_o + 460}{100} \right)^4 \right] \quad (1)$$

$$\left(\frac{q}{A}\right)_{\text{conv.}} = 0.3 \left[\frac{T_b - T_o}{2} \right]^{1.25} \quad (2)$$

$$\left(\frac{q}{A}\right)_{\text{cond.}} = k_{\text{air}} \left[\frac{T_b - T_o}{\delta_{X_{\text{air}}}} \right] \quad (3)$$

$$\left(\frac{q}{A}\right)_b (\text{calc.}) = \left(\frac{q}{A}\right)_{\text{rad}} + \left(\frac{q}{A}\right)_{\text{conv.}} + \left(\frac{q}{A}\right)_{\text{cond}} \quad (4)$$

where $\left(\frac{q}{A}\right)$ = heat flux, $\frac{\text{Btu}}{\text{hr-ft}^2}$

ϵ = emissivity ($\epsilon_1 = \epsilon_2 = 0.8$, assumed)

T = temperature °F

P = reactor power, Mw

k = thermal conductivity, Btu/hr-ft-°F

δ_X = thickness, ft.

Subscripts rad ~ radiation

conv ~ convection

cond ~ conduction

b ~ outside surface of pressure vessel

o ~ inside surface of blast shield. ($T_o = 100 + 5P$, assumed)

Data taken during HRT operation include the flow rate, inlet and outlet temperatures of the cooling water through the blast shield cooling coils, and temperatures of the outside surface of the pressure vessel. If the cooling water to the blast shield is assumed to remove only the heat radiated from the pressure vessel surface, then a heat balance over this cooling water stream will give an experimental check on equations (1-4). The heat balance equation is

$$q = W C_p (T_{out} - T_{in}) \quad (5)$$

where w = flow rate of cooling water stream, lbs/hr

C_p = heat capacity of water, $\frac{\text{Btu}}{\text{lb} \cdot ^\circ\text{F}}$

q = transfer rate of heat, Btu/hr

Dividing q by A (the area of the outside surface of the pressure vessel) gives an experimental value of the heat flux from the pressure vessel outside surface which can be compared with $\frac{q}{A}$ as calculated from

equations (1-4). This comparison is made in Table I and Figure 1.

In Figure 1 the experimental values of $\frac{q}{A}$ are plotted as small

circles while the calculated values of $\frac{q}{A}$ are shown as a solid line.

As can be seen from Figure 1, most of the experimental values of $\frac{q}{A}$

lie above the calculated curve (especially for the higher power levels).

This is expected since the blast shield itself is also in the radiation field and is thus generating some heat which must also be removed by the cooling system. The heat generated in the blast shield has been estimated to be about 1-1/2% of that in the pressure vessel wall. (1)

Assuming the heat generation figures derived by R. D. Cheverton and E. R. Volk (3) (see equation 14) for the pressure vessel, the heat generated in the blast shield can be estimated by

$$\frac{Q}{A} \text{ blast shield} = 34 P \frac{\text{Btu}}{\text{hr ft}^2 (\text{surface area})}$$

where P = reactor power, megawatts

Table I

HRT Pressure Vessel Data From Run 16

Date	Time	Reactor Power	T _{Bl'k} Avg. Blkt. Temp.	T _b Avg. Pres. Vessel Surface Temp.	$\left(\frac{q}{A}\right)_b$ Heat Flux at Outside Surface of Pres. Vessel Calc. Exper.		$\left(\frac{q}{A}\right)_R$ Heat Flux at Inside Surface of Pres. Ves.		T _R Temp. of Inside Surface of Pressure Vessel	(T _R -T _b) Temp. Dif. Across Pres. Vessel Wall	σ _{th} Thermal Stress in Pressure Vessel	†
		Mw.	°C	°C	$\frac{\text{Btu}}{\text{hr ft}^2}$		$\frac{\text{Btu}}{\text{hr ft}^2}$		°C	°C	psi	
6-4-58	1200-1835	0.00001	260	242	1004	980	-1330		253.5*	11.5*	2200	
	1835-2400	0.3	240	228	893	980	- 270		238.7*	10.8*		
6-5-58	0-1200	0.325	233	226	875	950	- 170		232.2*	6.2*		
	1935-2400	0.233	230	224	857	950	- 430		228*	4.0*		
6-6-58	0-1200	0.28	235	228	892	920	- 330		233.5*	5.5*		
	1200-2400	0.36	240	235	943	920	- 160		239.3*	4.3*		
6-7-58	0-1235	0.29	240	234	934	960	- 360		238.3*	4.3*		
	1335-2400	0.64	238	235	941	960	+ 690		240.0	5.0	1800	⊙
6-8-58	0-1600	0.676	235	236	948		800		237.2	1.2		
	1900-2400	0.794	260	255	1104	1060	950		262.5	7.5		
6-9-58	0-1300	0.714	258	251.7	1080	1082	740		260.0	8.3		
	1300-2400	0.737	261	256.9	1125	1115	750		263.1	6.2		
6-10-58	0800-2400	0.652	275	267.8	1220	1320	360		276.2	8.4	2400	
6-11-58	0-0535	0.640	273.5	266.8	1210	1290	340		274.6	7.8		
	1100-2400	1.038	260.6	258.5	1140	1290	1640		264.3	5.8		
6-12-58	0-1400	1.008	255	250.3	1070	880	1640		258.7	8.4	2900	
	1400-2400	1.014	261.5	256.6	1119	1040	1590		265.1	8.5		
6-13-58	0-1200	1.040	258	255.7	1110	1240	1580		261.6	5.9		
	1435-2400	1.600	260	260.6	1145	1180	3340		266.4	5.8		
6-14-58	0-1200	1.560	261	259	1140	1420	3220		267.2	8.2	3600	
	1200-2400	1.504	260	260	1145	1420	3050		265.9	5.9		
6-15-58	0-1100	1.520	260	260	1145	1390	3090		266.0	6.0		
	1100-2400	2.089	261	264	1176		4780		269.3	5.3		
6-16-58	0-1400	2.017	260	262	1150		4590		268.1	6.1	3700	
	1400-2400	2.00	261									

Table I. (cont'd.)

Date	Time	Reactor Power	T _{Bl'k}	T _b	($\frac{q}{A}$) _b	($\frac{q}{A}$) _R	T _R	(T _R -T _b)	σ_{th}	†
6-17-58	0-1000	2.00	261	265.9	1145	4550	268.9	3.0		
	1200-2400	2.72	262	269.2	1215	6640	272.7	3.5		
6-18-58	0-1500	2.64	262	268.4	1210	6400	272.4	4.0	4100	
6-20-58	0-1500	3.258	263.9	274.4	1265	8200	276.3	1.9		
	1500-2400	3.19	262.6	274	1255	8000	274.8	0.8		
6-21-58	0-1315	3.17	264.3	274	1255	7950	276.5	2.5	4500	
6-22-58	0-2400	1.496	281.9	278.8	1315	2800	287.5	8.7		
6-23-58	0-0555	1.455	283.9	280.1	1330	2650	289.3	9.2	2000	
	1500-2400	3.38	269.6	280.2	1310	8510	282.4	2.2		
6-24-58	0-1200	3.39	268.2	278.7	1300	8560	281.1	2.4	5300	
	1200-2400	3.82	269.3	281.5	1320	9820	283.6	2.1		
6-25-58	0-0515	3.31	267.0	276.6	1290	8390	279.7	3.1		
	2215-2400	2.59	263.1	269.0	1210	6250	273.3	4.3		
6-26-58	0-1200	2.75	264.0	272.1	1240	1070	274.7	2.6		
	1200-2400	2.74	263.0	272.6	1250	1490	273.5	0.9		
6-27-58	0-0900	2.79	264.0	274.0	1260		274.8	0.8	4200	
	1200-1700	2.78	274.8	282.0	1340		285.3	3.3		
	1700-2400	3.59	275.0	286.5	1380		288.5	2.0		
6-28-58	0-1700	3.68	274.0	285.3	1370	1450	287.8	2.5	5200	
	1700-2400	4.22	272.8	288.9	1400	1420	288.2	-0.7		
6-29-58	0-2400	4.27	270.8	287.1	1380	1435	286.5	-0.6		
6-30-58	0-0600	4.32	271.0	288.3	1390	1530	286.9	-1.4	5200	
	0600-2400	4.58	271.0	290.0	1405	1560	287.6	-2.4		
7-1-58	0-1200	4.70	270.3	291.7	1420	1610	287.1	-4.6	5900	
	1200-2400	5.03	268.0	290.8	1420	1670	285.9	-4.9		
7-2-58	0400-1200	5.48	268.0	291.7	1410		287.4	-4.3	6600	
	1200-2400	5.28	267.5	290.0	1400		286.3	-3.7		
7-3-58	0-1200	5.44	267.3	291.5	1410		286.5	-5.0	6800	
	1500-1625	3.76	267.0	283.5	1350	1420	281.1	-2.4		

Note: Data taken for transition periods, i.e., from one power level to another or from one operating temperature level to another, were not used for this study. Therefore, there are some time gaps in the data.

* These numbers were calculated assuming $h_R = 120 \text{ Btu/hr.ft}^2.\text{°F.}$

† The thermal stress is calculated at the point in the vessel wall where it is a maximum. This occurred at the outside surface for all except the last three values listed.

At 5 Mw this represents a heat generation in the blast shield of 170 Btu/hr.ft², which is about 10% as great as the heat removed by radiation from the pressure vessel surface.

It should be noted that the heat loss to the cell atmosphere by the blast shield tends to partially counterbalance this effect. From Figure 1 it appears that the pressure vessel cooling system is performing satisfactorily.

Theoretical Heat Generation in Pressure Vessel

The amount of heat radiated from the outside pressure vessel surface is a function of the temperatures of this surface and the inside surface of the blast shield as indicated by equations (1-4). The heat generation, due to neutrons and gamma radiation inside the pressure vessel wall, however, is a function of reactor power level.

The differential equation for steady state heat conduction with an internal heat source is (for spherical coordinates)

$$\frac{d^2T}{dr^2} + \frac{2}{r} \frac{dT}{dr} + \frac{H(r)}{k} = 0 \quad (6)$$

where r = radius

$H(r)$ = internal heat source as function of radius.

If $H(r)$ can be expressed as a sum of exponential terms; i.e.

$$H(r) = \sum_{i=1}^n H_i P_i e^{-M_i(r-R)},$$

then equation (6) may be solved to give

$$\frac{dT}{dr} = \sum_{i=1}^n \frac{H_i P_i}{k m_i} e^{-m_i(r-R)} \left[1 + \frac{2}{m_i r} + \frac{2}{m_i^2 r^2} \right] + \frac{C_1}{r^2} \quad (7)$$

and

$$T = \sum_{i=1}^n - \frac{H_i P}{k m_i^3} \left[\frac{2 + m_i r}{r} \right] e^{-m_i(r-R)} - \frac{C_1}{r} + C_2 \quad (8)$$

where H_i , m_i = constants depending on heat generation.

C_1 , C_2 = arbitrary constants.

R = inside radius of pressure vessel wall.

The surface heat flux is expressed by Fourier's Law as

$$\frac{q}{A} = -k \frac{dT}{dr} \quad (9)$$

The above equations (6-9) may be solved (using the boundary conditions $T = T_b$ at $r = b$ and $T = T_R$ at $r = R$) to give the heat fluxes at the inside and outside surfaces of the pressure vessel as functions of power and temperature drop across the wall. These equations will be of the form

$$\left(\frac{q}{A} \right)_b = C_{H_1} P + C_{r_1} (T_R - T_b) \quad (10)$$

$$\left(\frac{q}{A} \right)_R = C_{H_2} P + C_{r_2} (T_R - T_b) \quad (11)$$

where C_{H_1} , C_{H_2} = constants depending only upon the internal heat sources

C_{r_1} , C_{r_2} = constants depending only upon the geometry of the system

Subscript b refers to radius of outside surface of pressure vessel

Subscript R refers to radius of inside surface of pressure vessel

R. D. Cheverton and E. R. Volk⁽³⁾ have derived equations, using a finite difference method, which express the heat fluxes at the inside and outside surfaces of the pressure vessel as functions of power level and temperature drop across the wall ($T_R - T_b$). They are

$$\left(\frac{q}{A}\right)_b = 873.9 P + 49.0 (T_R - T_b) \quad (12)$$

$$\left(\frac{q}{A}\right)_R = 1774 P - 64.5 (T_R - T_b) \quad (13)$$

The total heat generation based on one square foot of outside surface is then

$$\frac{Q}{A} = \left(\frac{q}{A}\right)_b + \frac{R^2}{b^2} \left(\frac{q}{A}\right)_R = 2220 P \quad \frac{\text{Btu}}{\text{hr ft}^2} \quad (14)$$

where Q = heat generation rate in the pressure vessel wall, ($= 222,000 \frac{\text{Btu}}{\text{hr Mw}}$)
 A = area of outside surface of pressure vessel, ($= 100 \text{ ft}^2$)
 P = reactor power, megawatts

For steady state operation, the heat generated in the vessel wall is removed in two ways: (1) by heat transfer to the blanket solution, and (2) by radiating heat to the blast shield where it is removed in the cooling coils.

The relative proportions of the heat removed by these methods vary with reactor power and operating temperature. If the reactor is just critical at essentially no power, the heat generated is negligibly small while the heat removed by the blast shield cooler is large in comparison. On the other hand, for five megawatt operation, the heat generated is about 11,500 Btu/hr ft². For a pressure vessel surface temperature of about 290°C the heat removed by the blast shield cooler is only about 1600 Btu/hr ft². The difference must be removed by the blanket solution. For steady state heat transfer the heat flux into the blanket solution from the inside surface of the pressure vessel can be estimated from

$$\left(\frac{q}{A}\right)_R = \frac{\frac{Q}{A} - \left(\frac{q}{A}\right)_b}{\frac{R^2}{b^2}} \quad (15)$$

where $\left(\frac{q}{A}\right)_R$ = heat flux at inside surface of pressure vessel, $\frac{\text{Btu}}{\text{hr ft}^2}$
 b = outside radius of pressure vessel (=2.875 ft.)
 R = inside radius of pressure vessel (=2,500 ft.)
 $\frac{R^2}{b^2}$ = correction for spherical geometry of system

Using equation (15) the heat flux at the inside pressure vessel surface is estimated as 13,100 Btu/hr ft² (at 5 Mw) which is about eight times the heat flux at the outside surface. Some inside and outside heat fluxes are given in Table I showing this relationship with power.

It appears from a study of the data based on heat generation given in equation (14) that below a power level of about 0.4 Mw the heat generated within the wall is removed entirely by the blast shield cooler. Above this power level heat is removed both by the blanket solution and the cooler. The point at which the blanket and blast shield cooler remove heat at equal rates occurs at a power level between 0.9 Mw and 1.0 Mw depending on the temperature of operation.

Temperature Drop Across Pressure Vessel Wall

An estimate of the temperature of the inside pressure vessel surface can be made using the average blanket temperature if a temperature drop across the blanket film can be computed. To do this an inside heat transfer coefficient is needed. If natural convection is assumed in the blanket then a heat transfer coefficient may be estimated as a function of temperature drop across the liquid film. This equation is (2)

$$h = 130 (\Delta T)_f^{1/3} \quad (16)$$

Then

$$\left(\frac{q}{A}\right)_R = h(\Delta T)_f = 130(\Delta T)_f^{4/3}$$

Solving for $(\Delta T)_f$

$$(\Delta T)_f = 0.026 \left(\frac{q}{A}\right)_R^{3/4} \quad (17)$$

where $(\Delta T)_f$ = temperature drop across blanket liquid film, °F

Thus the temperature of the inside pressure vessel surface, T_R , is

$$T_R = T_{Bl'k} + (\Delta T)_f \quad (18)$$

where $T_{Bl'k}$ = average of the measured inlet and outlet blanket temperatures.

The temperature drop across the vessel wall is then

$$(\Delta T)_w = T_R - T_b \quad (19)$$

Values of the temperature drop across the vessel wall, computed in this manner using equations (14), (15) and (17), are given in Table I and Figure 2.

Using curves derived by R. D. Cheverton⁽³⁾, the thermal stress can be estimated for various power levels and temperature differentials across the pressure vessel. This was done for several different power levels using the heat generation values given by equations (12), (13) and (14). The results are presented in Table I. The thermal stress was calculated for the highest $(\Delta T)_w$ values shown in the Table at each power level as this gives the most pessimistic stress values. It appears that the pressure vessel is in no danger from this source at power levels up to the design power of 5 Mw.

To compare the $(\Delta T)_w$ values of Figure 2 with theoretical values an expression may be derived from Cheverton's⁽³⁾ equations which relates $(\Delta T)_w$ and P. To do this the outside surface temperature, T_b , may be written

$$T_b = T_{Bl'k} + (\Delta T)_f - (\Delta T)_w$$

The temperature drop across the blanket solution surface film is found by substituting equation (13) in equation (17).

$$(\Delta T)_f = 0.026 \left[1774 P - 64.5 (\Delta T)_w \right]^{0.75} \quad (20)$$

Equation (12) may be written

$$P = \frac{1}{873.9} \left[\left(\frac{q}{A} \right)_b - 49.0 (\Delta T)_w \right] \quad (12a)$$

Figure 2

Temperature Drop Across
HRT Pressure Vessel Wall
(based on theoretical heat
generation rates in vessel)

Substituting for $\frac{q}{A}$ using equations (1-4) and (20) results in an expression relating $(\Delta T)_b$ and power with blanket temperature as parameter.

$$P = \frac{1}{873.9} \left\{ 0.1154 \left\{ \frac{[T_{Bl'k} + 0.026 [1774P - 64.5(\Delta T)_b]^{0.75} - (\Delta T)_b + 460]}{100} \right\}^4 - \left[\frac{560 + 5P}{100} \right]^4 \right\} + 0.3 \left\{ \frac{T_{Bl'k} + 0.026 [1774P - 64.5(\Delta T)_b]^{0.75} - (\Delta T)_b - 100 - 5P}{2} \right\}^{1.25} + 0.16 \left\{ T_{Bl'k} + 0.026 [1774P - 64.5(\Delta T)_b]^{0.75} - (\Delta T)_b - 100 - 5P \right\} - 49(\Delta T)_b \quad (21)$$

Equation (21) is plotted on Figure (2) as a solid line for a blanket temperature of 260°C. Over the power range 0-5 Mw. the line representing equation (21) is nearly straight. The slope of this line has a greater negative value than the slope of the dashed line drawn through the data points. This would indicate a heat generation rate less than that predicted by equation (14). This appears reasonable as calculations of this type are usually conservative.

An examination of equation (21) will show that the slope of the theoretical curve is dependent mainly upon the heat generation sources within the vessel wall. This suggests a method by which the actual heat generation can be evaluated from the data. The procedure by which this can be accomplished is outlined in Appendix A. Results of this study are presented in Table II and Figure 3. The indicated heat generation rate in the pressure vessel wall is

$$\frac{Q}{A} = 1080 P \frac{\text{Btu}}{\text{hr. ft}^2 (\text{outside surface})} \quad (22)$$

$$\text{where } Q \approx 108,000 \frac{\text{Btu}}{\text{hr Mw.}}$$

The heat flux equations now become

$$\left(\frac{q}{A} \right)_b = 426.1 P + 49.0 (\Delta T)_b \quad (23)$$

$$\left(\frac{q}{A} \right)_R = 865.0 P - 64.5 (\Delta T)_b \quad (24)$$

Table II

Calculated Data for HRT Pressure Vessel on Heat Generation Rate Given in Equation 22

Date	Time	Reactor Power	$\left(\frac{q}{A}\right)_R$ Heat Flux at Inside Surface	T_R Temp. of Inside Surface (Eq.18)	$(T_R - T_b)$	$(T_R - T_b)$ theor. (From Fig.4)	$(T_{Bl'k} - T_b)$ (From Table I)	Required $(\Delta T)_f$	h_f Blanket Film Heat Transfer Coefficient
		Mw	$\frac{\text{Btu}}{\text{hr ft}^2}$	$^{\circ}\text{C}$	$^{\circ}\text{C}$	$^{\circ}\text{C}$	$^{\circ}\text{C}$	$^{\circ}\text{F}$	$\frac{\text{Btu}}{\text{hr.ft}^2.^{\circ}\text{F}}$
6-4-58	1200-1835	0.00001	-1328			11.75	18	-11.25	118*
	1835-2400	0.3	- 752.8			8.9	12	- 5.58	135
6-5-58	0-1200	0.325	- 693.3			8.3	7	2.34	-296
	1935-2400	0.233	- 800.9			8.5	6	4.5	-176
6-6-58	0-1200	0.28	- 780.0			8.7	7	3.06	-255
	1200-2400	0.36	- 733.2			8.6	5	6.48	-113
6-7-58	0-1235	0.29	- 821.3			9.0	6	5.4	-152
	1335-2400	0.64	- 330.5			7.3	3	7.74	- 43
6-8-58	0-1600	0.676	- 288.3			6.9	-1	14.2	- 20
	1900-2400	0.794	- 326.1			8.3	5	5.94	- 55
6-9-58	0-1300	0.714	- 408.7			8.6	6.3	4.14	- 99
	1300-2400	0.737	- 435.3			8.7	4.1	8.28	- 53
6-10-58	0800-2400	0.652	- 682.4			10.3	7.2	5.58	-122
6-11-58	0-0535	0.640	- 686.4			10.2	6.7	6.3	-109
	1100-2400	1.038	25.1	260.6	2.1	7.3	2.1	9.36	- 2.7
6-12-58	0-1400	1.008	24.6	255.2	4.9	7.1	4.7	4.32	5.7
	1400-2400	1.014	- 31.6	261.5	4.9	7.6	4.9	4.86	- 6.5
6-13-58	0-1200	1.040	17.5	258.1	2.4	7.2	2.3	8.82	2.0
	1435-2400	1.600	771	262.1	1.5	4.9	-0.6	9.9	78
6-14-58	0-1200	1.560	721	263.0	4.0	5.2	2.0	5.76	125
	1200-2400	1.504	634	261.8	1.8	5.3	0	9.54	66
6-15-58	0-1100	1.520	658	261.9	1.9	5.25	0	9.45	70
	1100-2400	2.089	1429	264.4	0.4	2.9	-3	10.62	135
6-16-58	0-1400	2.017	1360	263.3	1.3	3.2	-2	9.36	145
6-17-58	0-1000	2.00	1343	264.2	-1.7	3.3	-4.9	14.76	91
	1200-2400	2.72	2280	266.8	-2.4	0.2	-7.2	13.32	171

Table II (cont'd.)

Date	Time	Reactor Power	$\left(\frac{q}{A}\right)_R$	T_R	$(T_R - T_b)$	$(T_R - T_b)_{\text{theor.}}$	$(T_{Bl,k} - T_b)$	Required $(\Delta T)_f$	h_f
6-18-58	0-1500	2.64	2171	266.6	-1.8	0.6	-6.4	12.6	172
6-20-58	0-1500	3.258	2982	269.7	-4.7	-1.8	-10.5	15.66	190
	1500-2400	3.19	2897	268.3	-5.7	-1.5	-11.4	17.82	163
6-21-58	0-1315	3.17	2870	270.0	-4.0	-1.4	-9.7	14.94	192
6-22-58	0-2400	1.496	398	283.2	4.4	7.2	3.1	7.38	54
6-23-58	0-0555	1.455	319	285.0	4.9	7.6	3.8	6.84	47
	1500-2400	3.38	3096	275.6	-4.6	-1.9	-10.6	15.66	198
6-24-58	0-1200	3.39	3124	274.3	-4.4	-2.0	-10.5	15.3	204
	1200-2400	3.82	3708	276.2	-5.3	-3.7	-12.2	15.3	242
6-25-58	0-0515	3.31	3052	272.9	-3.7	-1.8	-9.6	14.04	217
	2215-2400	2.59	2100	267.6	-1.4	0.9	-5.9	12.24	172
6-26-58	0-1200	2.75	2285	268.8	-3.3	0.4	-8.1	15.3	149
	1200-2400	2.74	2258	267.7	-4.9	0.3	-9.6	17.8	127
6-27-58	0-0900	2.79	2319	268.8	-5.2	0.2	-10.0	18.36	126
	1200-1700	2.78	2192	279.4	-2.6	1.2	-7.2	15.1	145
	1700-2400	3.59	3306	281.3	-5.2	-2.2	-11.5	16.74	197
6-28-58	0-1700	3.68	3449	280.5	-4.8	-2.7	-11.3	15.5	223
	1700-2400	4.22	4173	280.3	-8.6	-5.0	-16.1	20.0	209
6-29-58	0-2400	4.27	4279	278.5	-8.6	-5.3	-16.3	19.8	216
6-30-58	0-0600	4.32	4334	278.7	-9.6	-5.8	-17.3	20.7	209
	0600-2400	4.58	4682	279.2	-10.8	-6.8	-19.0	22.0	213
7-1-58	0-1200	4.70	4837	278.7	-13.0	-7.3	-21.4	25.4	191
	1200-2400	5.03	5301	277.0	-13.8	-8.9	-22.8	25.0	212
7-2-58	0400-1200	5.48	5964	277.8	-13.9	-10.8	-23.7	23.2	257
	1200-2400	5.28	5692	277.0	-13.0	-10.0	-22.5	22.5	253
7-3-58	0-1200	5.44	5907	277.0	-14.5	-10.6	-24.2	24.5	241
	1500-1625	3.76	3587	273.7	-9.8	-3.6	-16.5	23.2	154

* This value agrees well with the value of $h_f = 114$, estimated for zero power level on page 18 of this study.

The theoretical equation relating $(\Delta T)_v$ and P now becomes

$$P = \frac{1}{426.1} \left\{ 0.1154 \left\{ \left[\frac{T_{Bl'k} + 0.026(865P - 64.5\Delta T_v)^{0.75} - \Delta T_v + 460}{100} \right]^4 - \left[\frac{560 + 5P}{100} \right]^4 \right\} + 0.3 \left\{ \frac{T_{Bl'k} + 0.026(865P - 64.5\Delta T_v)^{0.75} - \Delta T_v - 100 - 5P}{2} \right\}^{1.25} + 0.16 \left\{ T_{Bl'k} + 0.026(865P - 64.5\Delta T_v)^{0.75} - \Delta T_v - 100 - 5P - 49\Delta T_v \right\} \dots \right\} \quad (25)$$

The heat generation values given in equations (22-25) are based upon the assumption of natural convection heat transfer (as given by Equation (16)) between the blanket solution and the metal surface of the pressure vessel. The assumption of forced convection heat transfer is considered in Appendix B. For this heat transfer mechanism the overall heat generation rate is estimated to be

$$\left(\frac{Q}{A} \right)_{\text{forced conv.}} = 885 P \frac{\text{Btu}}{\text{hr.ft}^2(\text{surface area})} \quad (43)$$

$$\text{where } Q_{\text{forced conv}} \approx 88,500 \frac{\text{Btu}}{\text{hr Mw.}}$$

The actual heat generation should lie somewhere between the values indicated in equations (22) and (43), probably nearer the value given in equation (22).

It should be emphasized here that the value of T_B , the temperature of the inside wall of the pressure vessel, was found from the estimated heat flux which depends primarily upon the heat generation rate in the wall. The value of ΔT_v , the temperature drop across the vessel wall, is therefore dependent upon the heat generation rate. The heat generation in turn was estimated, using the values of ΔT , by the method outlined in Appendix A. Thus, the heat generation is not completely independent of the ΔT_v calculations, and the possibility exists that one may be misled in his conclusions. However, it is felt that these calculations are sufficiently independent for valid estimates to be made. In order to obtain a more conclusive check on the overall heat generation rate as well as the form which the heat sources take in the metal wall, it is recommended that a complete temperature profile measurement be attempted on the replacement reactor pressure vessel. If the complete temperature profile cannot be made, then provision for the measurement of the inside surface temperature is recommended.

Heat Transfer Film Coefficient

It will be noticed that the theoretical curve of Figure 3 falls above the least squares line through the data. This would indicate a higher surface temperature than was calculated using the heat transfer coefficient of equation (16). In order to determine the temperature drop, ΔT_f , across the blanket film which would be required for the ΔT_f values to fall on the theoretical curve, the following equation is used:

$$\Delta T_f = \Delta T_{f(\text{theoretical})} - (T_{Bl'k} - T_b) \quad (26)$$

To determine $\Delta T_{f(\text{theoretical})}$, equation (25) is plotted for several different blanket temperatures in Figure 4. From this plot a theoretical value of ΔT_f can be read for each data point. The heat transfer coefficient, h , which would be required to give this film temperature drop can then be estimated from

$$h = \frac{(q/A)_R}{\Delta T_f} \quad (27)$$

Values of h are given in Table II.

At power levels below 2 Mw. there appear to be no definite relation between h and ΔT_f . Above this power level, however, the value of h increases as ΔT_f increases. Natural convection heat transfer has been postulated as the mechanism by which heat is transmitted from the inside surface of the pressure vessel to the blanket solution. To check this postulation a log-log plot of h versus ΔT_f was made (Figure 5). The data appear widely scattered. However, small errors in the measurement of $T_{Bl'k}$ and T_b are relatively large when compared to ΔT_f and the scatter of this plot might be expected when one considers the difficulties involved in measuring temperatures in a reactor system of this type. A least squares line was calculated for the data of Figure 5. The equation of this line is

$$h = 56.9 \Delta T_f^{0.411} \quad (28)$$

It should be remembered that the differences between the calculated values of ΔT_f and the values read from Figure 4 are only a few degrees (0 to 10°C) as can be seen in Table II. A consistent error in temperature measurement would have a considerable effect on the constants (both coefficient and exponent) in equation (28). Such an error may exist in the present HRT system. Considering the blanket temperatures above, for example, the raw data taken on June 4, at zero power show the inlet solution temperature reading 2°C above the outlet solution

Figure 4

Theoretical Temperature Drop
Across HRT Pressure Vessel Wall
(Plot of equation 25)

temperature. Again on the following day the power was reduced to essentially zero, and the blanket inlet temperature read between 1°C and 2°C above the outlet temperature. At the temperature of operation during these periods of zero power, the heat loss from the outside surface of the pressure vessel would account for only about 1/2°C. It is likely then that at least one of these measurements is consistently off. Since the temperature values given in Table I for blanket solutions are averages of the measured inlet and outlet temperatures it is possible that a consistent error of 1°C or 2°C exists in the value of T_{Blank} . To show the effect of errors of these magnitudes, values of blanket solution temperature 1°C below, 1°C above, and 2°C above the values given in Table I were assumed. Values for ΔT_f and h , which would force a fit with theoretical equation (25) were then estimated using the method described previously. The least squares line through a log-log plot of h_f versus ΔT_f was calculated for each case. The results are:

For a blanket temperature 1°C below values given in Table I

$$h = 31.5 \Delta T_f^{0.563} \quad (28a)$$

For a blanket temperature 1°C above values given in Table I

$$h = 105 \Delta T_f^{0.246} \quad (28b)$$

For a blanket temperature 2°C above values given in Table I

$$h = 190 \Delta T_f^{0.081} \quad (28c)$$

One would infer from the wide scatter of the data as seen in the Figure 5 plot, that a small change in the independent variable, ΔT_f , might have a pronounced effect on the least squares line equation. This is borne out by the equations given above. It should be obvious, then, that equations (28) should not be used to predict a heat transfer coefficient at the inside surface of the pressure vessel. The only conclusion that has been drawn, on the basis of the data available from Run 16, concerns the mechanism by which heat is transmitted across the blanket film. The exponent of ΔT_f in equation (28) compares fairly well with that of equation (16) which was used in computing ΔT_f .

$$h = 130 \Delta T_f^{1/3} \quad (16)$$

This indicates that heat may be transferred from the vessel wall to the blanket solution mainly by natural convection. Heat transfer by forced convection alone would give values for h which are not dependent upon ΔT_f .

Since natural convection heat transfer was postulated initially as the mechanism by which heat is transferred across the blanket solution film, it may be argued that if forced convection were assumed to begin with, the results might have indicated a forced convection mechanism rather than a natural convection mechanism. To check this possibility forced convection was assumed. The results indicate an even greater dependence of the film coefficient of heat transfer, h , on the temperature drop across the film, ΔT_f . The forced convection method is outlined in Appendix B and the least squares line through a log-log plot of h_f versus ΔT_f is expressed by

$$h = 12.0 \Delta T_f^{0.681} \quad (47)$$

Equation (47) can thus be offered as added evidence that the mechanism by which heat is transferred is principally natural convection.

As can be seen from Table II, the heat transfer coefficients calculated for power levels below 2 Mw vary from -296 to 125 Btu/hr ft² °F. They are not included in the plot of Fig. 5 because at these power levels the heat fluxes are small and any error here would be magnified in the heat transfer coefficient values. This, along with the errors inherent in the temperature measurements, would make any analysis which is based on these data, of doubtful validity.

Also below about one megawatt during normal operation the heat at the inside surface of the pressure vessel flows from the blanket solution into the metal wall. The mechanism for heat transfer in this case would certainly be different from that at the higher power levels since the direction in which the fluid is being pumped is opposite to the flow that would be induced by natural convection. The conclusion that is reached is that the data are not good enough to estimate the film coefficients in this region.

The scatter of data as seen in Fig. 5 is such that any conclusions as to heat transfer mechanism should be considered preliminary at best. The conclusions drawn in this section are based on 29 data points, calculated for reactor power levels greater than 2 Mw. Many more data are needed if the conclusions indicated are to be accepted. It is recommended, therefore, that data taken in succeeding runs of the HRT, be analyzed in a manner similar to that presented in this study.

Operation of Reactor Without Cooling Water in Blast Shield Coils

On July 3, (just before shutdown) the water to the cooling coils on the blast shield was cut off. The reactor was operating at 3.76 Mw. at this time. The data taken is listed in Table III.

Table III
Data Taken with Blast Shield Cooling Water Off

Time	Cooling Water	$T_{Bl'k}, ^\circ C$	$T_b, ^\circ C$	Cooling Water Temperature, $^\circ C$	
				In	Out
1415	on	267	281.5	30	38.5
1615	on	268	283	30	38.5
1645	off*	267	283.6	30	72.5
1705	off	267	283.9	30	95.5
1745	off	267	285	30	116.5
1815	off	267	283.9	30	115.5
1840	off	267	283.4	30	116.3
1930	off	268	284.2	30	115.5
1955	off	268	284.6	30	116
2135	off	274	280.9	29	116.2

* Cooling Water turned off at 1625

The purpose of this was to see if a value for (ΔT) could be obtained for the condition where the outside of pressure vessel is insulated. In order to attain an insulated pressure vessel surface in the present system the blast shield must be heated to approximately the same temperature as the pressure vessel. To do this there are two heat sources available: (1) heat generation in the blast shield metal itself which amounts to about 130 Btu/hr.ft² at 3.76 Mw. and (2) the heat input due to radiation from the surface of the pressure vessel (this varies from about 1360 Btu/hr.ft² for a blast shield temperature of 100° F to zero when the blast shield and pressure vessel are at the same temperature). From a rough approximation assuming these two heat sources, water initially in the coils at 95° F, about 82 pounds of steel per sq. ft. of blast shields surface, and no heat loss from the blast shield to the surroundings it will take about one hour to boil the water dry. This checks the experimental data given above. A second hour is required for the temperature of the steel to rise to 300° F. As the blast shield temperature rises, however, the heat

losses (by radiation to the cell walls and convection to the cell atmosphere) become more important. In fact, it is doubtful if the temperature of the blast shield was much above 116°C , the temperature registered on the cooling water outlet line, at the end of the run. It would probably take about 24 hours or longer for the blast shield to reach its maximum temperature after turning off the cooling water, and this maximum may be well below the pressure vessel surface temperature. It is doubtful then that conditions approaching an insulated pressure vessel surface can be attained in the present system.

Maximum Inside Surface Temperature

There is at present no means of measuring the temperature of the inside surface of the pressure vessel. The reliability of the values of T_R given in Table I is only as good as equation (14) and (16). Since the blanket solution of the HRT now contains uranium there is some concern over volume heat sources in the liquid next to the pressure vessel surface. It may be useful, therefore, to calculate the maximum temperature that the inside surface could attain as a function of the outside surface temperature, T_b , which is measured experimentally. This is done assuming normal operating conditions with cooling water running through the blast shield cooling coils.

The relation between the heat flux at the outside surface, $\left(\frac{q}{A}\right)_b$, and the slope of the temperature distribution at the outside surface of the metal wall is given by Fourier's Law of heat conduction.

$$\left(\frac{q}{A}\right)_b = -k \left(\frac{dT}{dr}\right)_b \quad (29)$$

In order to obtain the upper limit of the temperature distribution throughout the vessel wall for a given surface temperature, T_b , the assumption of no heat generation within the metal wall is made. Then the heat flux at any radius r is given by

$$\left(\frac{q}{A}\right)_r = \frac{A_b}{A_r} \left(\frac{q}{A}\right)_b = \frac{b^2}{r^2} \left(\frac{q}{A}\right)_b \quad (30)$$

The slope of the temperature distribution at any radius is then

$$\left(\frac{dT}{dr}\right)_r = -\frac{1}{k} \left(\frac{q}{A}\right)_r = -\frac{b^2}{kr^2} \left(\frac{q}{A}\right)_b \quad (31)$$

Separating the variables

$$dT = -\frac{b^2}{k} \left(\frac{q}{A}\right)_b \frac{1}{r^2} dr \quad (32)$$

Since the value of thermal conductivity, k , is different in the cladding from that of the carbon steel, equation (32) is integrated over two ranges of r . For the carbon steel

$$\int_{T_b}^T dT = -\frac{b^2}{k_c} \left(\frac{q}{A}\right)_b \int_b^r \frac{dr}{r^2} \quad \text{for } a \leq r \leq b$$

or

$$T = T_b + \frac{b^2 \left(\frac{q}{A}\right)_b}{k_c} \left[\frac{1}{r} - \frac{1}{b} \right] \quad (33)$$

where a = radius of the interface surface between the cladding and the carbon steel (= 2.542 ft)

k_c = thermal conductivity of carbon steel (= 23.8 Btu/hr.ft. $^{\circ}$ F)

Similarly for the stainless steel cladding (for $R \leq r \leq a$)

$$T = T_a + \frac{b^2}{k_s} \left(\frac{q}{A}\right)_b \left[\frac{1}{r} - \frac{1}{a} \right] \quad (34)$$

where k_s = thermal conductivity of stainless steel (= 11.2 $\frac{\text{Btu}}{\text{hr.ft.}^{\circ}\text{F}}$)

Equations (33) and (34) give the temperature distribution throughout the pressure vessel wall as a function of outside surface heat flux for the case of no internal heat generation. For a given surface temperature, T_b , and heat flux, $\left(\frac{q}{A}\right)_b$, this will then provide an

upper limit for the actual temperature distribution which, while the reactor is operating at power, would include the effect of a volume heat source within the metal wall.

Equations (33) and (34) may be used to find the maximum temperature difference across the vessel wall, $(T_R - T_b)_{\max}$. This is given by

$$(T_R - T_b)_{\max} = b^2 \left(\frac{q}{A}\right)_b \left\{ \frac{1}{k_c} \left[\frac{1}{a} - \frac{1}{b} \right] + \frac{1}{k_s} \left[\frac{1}{k} - \frac{1}{a} \right] \right\} \quad (35)$$

$(T_R - T_b)_{\max}$ is plotted versus T_b in Figure 6.

From T_b , the temperature reading on the outside surface of the pressure vessel, a value of $(T_R - T_b)_{\max}$ may be read from Figure 6. Adding this to T_b gives $T_{R, \max}$, the maximum temperature that the inside surface of the metal wall could attain regardless of the power level. If the power level is above 1 to 2 Mw. the heat flux at the inside surface of the pressure vessel is from the wall into the blanket solution. Thus, the temperature of the blanket solution next to the pressure vessel wall would be equal to or less than T_R . Figure 6 may, therefore, prove useful in helping to determine a maximum operating temperature of the blanket solution below which there would be no danger of phase separation in the uranyl sulfate solution next to the pressure vessel wall.

When the reactor is operating at power, the actual value for $(T_R - T_b)$ will be smaller than that given in Figure 6. It will approach this value only when the power level approaches zero. It should be pointed out that this $(\Delta T)_r$ may actually exist during the delay time before dumping; i.e., that period during which the dissolved radiolytic gases are being recombined in preparation for the dump to low pressure. A consideration of the thermal stress during this period shows that it will not exceed allowable limits. For a period following 5 Mw. operation and with T_b reading 300°C , the thermal stress would be only 3400 psi.

Figure 6

Temperature Difference Across Pressure
Vessel Wall Assuming No Heat Generation
In Wall

T_R = inside surface temperature of pressure vessel

T_b = outside surface temperature of pressure vessel

HRT at Zero Power

It may be interesting to note that the temperature drop for zero power (HRT data for June 4) shows that the inside surface temperature is 253.5°C which compares with 260°C for the blanket solution. In this case, since no heat was being generated in the wall, the heat flux at the inside surface was

$$\left(\frac{q}{A}\right)_R = \frac{b^2}{R^2} \left(\frac{q}{A}\right)_b = 1330 \frac{\text{Btu}}{\text{hr ft}^2}$$

The heat transfer coefficient between the blanket solution and the vessel surface is then

$$h_f = \frac{1330 \frac{\text{Btu}}{\text{hr ft}^2}}{(260-253.5)^\circ\text{C} (1.8 \frac{^\circ\text{F}}{^\circ\text{C}})} = 114 \frac{\text{Btu}}{\text{hr ft}^2 \cdot ^\circ\text{F}}$$

The heat transfer coefficient predicted by equation (16) is

$$h = 130 \left[1.8(260-253.5) \right]^{1/3} = 290 \frac{\text{Btu}}{\text{hr.ft}^2 \cdot ^\circ\text{F}}$$

Although this is not a good check it must be realized that in this case, since the wall was cooler than the blanket solution, the tendency was for the fluid next to the wall to move downward. However, the blanket solution was moving upward due to the direction in which it was being pumped. This would inhibit the movement of the layer next to the wall which would tend to decrease the heat transfer coefficient. For power levels greater than about 1/2 to 1 megawatt the heat flux is from the vessel wall into the blanket and the natural convection tendency is to rise and equation (16) should be applicable.

Appendix A

Method of Computing Heat Generation Rate

Equation (21) relates power and temperature drop across the pressure vessel wall with blanket temperature as a parameter. It is derived from equation (12a)

$$P = \frac{1}{873.9} \left[\left(\frac{q}{A} \right)_b - 49.0(\Delta T)_f \right] \quad (12a)$$

The value of $\left(\frac{q}{A} \right)_b$ is computed from equations (1-4) which involve T_b , the temperature of the outside surface of the pressure vessel. T_b may, however, be written in terms of blanket temperature, $T_{Bl'k}$, $(\Delta T)_f$, and $(\Delta T)_f$. ΔT_f may be expressed as a function of $(\Delta T)_f$ using equations (13) and (17). The constant coefficient of equation (12a), $1/873.9$, is dependent only upon the heat sources within the pressure vessel wall as indicated by equation (10) and (11). The temperature is relatively small and, therefore, a plot of $(\Delta T)_f$ (theoretical) versus power will give an essentially straight line, the slope of which is dependent upon the heat source constant.

The experimental values of $(\Delta T)_f$ are found in the following manner:

- (1) Calculate the heat flux, $\left(\frac{q}{A} \right)_b$, from the outside surface of the pressure vessel using equations (1-4).
- (2) Calculate the overall heat generation rate in the metal wall using equation (14).
- (3) Calculate the heat flux, $\left(\frac{q}{A} \right)_R$, from the inside surface of the vessel using equation (15).
- (4) Estimate $(\Delta T)_f$, the temperature drop across the blanket solution film at the inside surface of the vessel, using eq. (17).
- (5) Add $(\Delta T)_f$ to the measured blanket temperature, $T_{Bl'k}$, and subtract the measured outside surface temperature T_b , to get $(\Delta T)_f$.

If the assumption is made that the mechanism of heat transfer at the inside wall is that indicated by equation (16), then a line drawn through the plot of the experimental values of $(\Delta T)_f$ versus power should have the same slope as equation (21). If the slopes are not equal then a new value for the heat source constant of equation (12a) is assumed and the process is repeated. For each new heat source constant of equation (12) the heat source constants of equations (13) and (14) must be adjusted proportionally. This is a simple matter as the relationship is linear.

Equation (21) is plotted easily by trial and error and the slope of the resulting straight line is taken from the plot. The slope of the "best" straight line through the experimental data points is found by the method of least squares. Let the equation of the least squares line be

$$\Delta T = s P = I \quad (36)$$

where s = slope

I = intercept with (ΔT) axis

The deviation of any point, i , from this line may be represented

$$\text{Deviation} = (\Delta T)_i - s P_i - I$$

Let σ = sum of squares of the deviation; i.e.,

$$\sigma = \sum_{i=1}^n \left[(\Delta T)_i - s P_i - I \right]^2 \quad (37)$$

where n = number of experimental points available.

To find the values of s and I corresponding to the minimum value of σ let

$$\frac{\partial \sigma}{\partial s} = 0 = \sum_{i=1}^n 2 \left[(\Delta T)_i - s P_i - I \right] (-P_i) \quad (38)$$

and

$$\frac{\partial \sigma}{\partial I} = 0 = \sum_{i=1}^n 2 \left[(\Delta T)_i - s P_i - I \right] (-1) \quad (39)$$

Rewriting

$$s \sum_{i=1}^n P_i^2 + I \sum_{i=1}^n P_i = \sum_{i=1}^n (\Delta T)_i P_i$$

and

$$s \sum_{i=1}^n P_i + I n = \sum_{i=1}^n (\Delta T)_i$$

Solving for s and I

$$s = \frac{n \sum_{i=1}^n (\Delta T)_i P_i - \sum_{i=1}^n (\Delta T)_i \sum_{i=1}^n P_i}{n \sum_{i=1}^n P_i^2 - \left[\sum_{i=1}^n P_i \right]^2} \quad (40)$$

$$I = \frac{\sum_{i=1}^n P_i^2 \sum_{i=1}^n (\Delta T)_i - \sum_{i=1}^n P_i \sum_{i=1}^n (\Delta T)_i P_i}{n \sum_{i=1}^n P_i^2 - \left[\sum_{i=1}^n P_i \right]^2} \quad (41)$$

Appendix B

Forced Convection Assumption

In order to see if the assumption of forced convection rather than natural convection would lead to different results the assumption is made that

$$h_f = 200 \frac{\text{Btu}}{\text{hr ft}^2 \text{ } ^\circ\text{F}} = \text{constant} \quad (42)$$

(any value of h_f could be chosen as the slope of the line relating ΔT and P is not dependent upon this).

The temperature drop across the vessel wall (ΔT_w) (exp'l.) is then computed using equations (15), (18), (19), (22), and (42) in the same manner as was done before when natural convection was assumed. The theoretical curve was then calculated from equations (23) and (24). In this case, the slope of the least squares line was different from the slope of the theoretical line indicating that the predicted overall heat generation rate would be different if forced convection were the postulated mechanism.

The overall heat generation was then estimated by the method outlined in Appendix A. The results indicate a heat generation rate (based on one-ft² of outside pressure vessel surface and) of

$$\left(\frac{Q}{A}\right)_{\text{forced conv.}} = 885 P \frac{\text{Btu}}{\text{hr}} \quad (43)$$

The heat flux equations for this rate are

$$\left(\frac{q}{A}\right)_b = 349.1 P + 49(T_R - T_b) \quad (44)$$

$$\left(\frac{q}{A}\right)_R = 708.8 P - 64.5(T_R - T_b) \quad (45)$$

The theoretical relationship between power and ΔT then becomes

$$\begin{aligned}
 P = & \frac{1}{349.1} \left\{ 0.1154 \left[T_{B1,k} + 0.005(708.8P - 64.5\Delta T) - \Delta T_{\checkmark} + 460 \right]^4 - \left[\frac{560+5P}{100} \right]^4 \right\} \\
 & + 0.3 \left\{ \frac{T_{B1,k} + 0.005(708.8P - 64.5\Delta T_{\checkmark}) - \Delta T_{\checkmark} - 100 - 5P}{2} \right\}^{1.25} \\
 & + 0.16 \left\{ T_{B1,k} + 0.005(708.8P - 64.5\Delta T) - \Delta T_{\checkmark} - 100 - 5P \right\} - 49 \Delta T_{\checkmark} \dots \quad (46)
 \end{aligned}$$

The value of $\Delta T_{\checkmark}$ (theoretical) can then be found from equation (46). From this the values of ΔT_f and h_f , which would be required to force a fit of equation (46), can be calculated. The least squares line through the log-log plot of h_f versus ΔT_f is given by

$$h_f = 12.0 (\Delta T_f)^{0.681} \quad (47)$$

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